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FLEX-GEARS

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Abstract:

Flex-Gears are being developed as an alternative to brushes and slip rings to conduct electricity across a rotating joint. Flex-Gears roll in the annulus of sun and ring gears for electrical contact while maintaining their position by using a novel application of involute gears. A single Flex-Gear is predicted to transfer up to 2.8 Amperes, thereby allowing a six inch diameter device, holding 30 Flex-Gears, to transfer over 80 Amperes. Semi-rigid Flex-Gears are proposed to decrease Flex-Gear stress and insure proper gear meshing.

Nomenclature:

L	circumferential length of rolling contact of flexible ring
N_p	number of teeth on planet gear (integer)
R	radius of pitch circle
V_t	relative velocity of mating gear teeth at point of contact and tangential to the surfaces of contact
b	radial thickness of flexible ring
c_r	contact ratio along the line of action or pressure line
h	tooth height
p	circular pitch, distance between tooth centers
r_b	radius of base circle
t	tooth thickness along the pitch circle
λ	characteristic number, ratio of Flex-Gear contact to that of a flexible ring
ϕ	pressure angle

Note to the reader:

The second and third sections of this paper, Original Flex-Gears and Rolling-Flex-Gears, portray the evolution of Flex-Gears and can be skipped without risking understanding of Flex-Gears.

Introduction:

There are many applications that transfer electricity between moving parts. The Conduction of electricity across *continuously* rotating joints is especially important to motor commutation and robot arm joints. Brushes and slip rings are commonly used for this purpose, but wear severely in vacuum environments such as space. Slip rings wear radially much less than brushes, but volumetrically wear the same [Holm, 1967]. Brush and slip ring wear is brought on by the combination of sliding, pressure, and the flow of electricity [Holm, 1967]. Brushless motors are often used in space applications; however, they exhibit much less stall torque than brush motors.

Roll rings were developed as an alternative to brushes and slip rings to conduct electrical power across a continuously rotating joint with little wear [Sperry Corporation, 1981]. The Roll Ring device shown in Figure 1 employs one or more flexible *planets* to conduct electricity between two concentrically rotating conductors called *sun* and *ring* [Porter, 1985]. The Roll Ring device incurs much less wear than brushes and slip rings and has the potential of transferring an electrical signal with little noise.

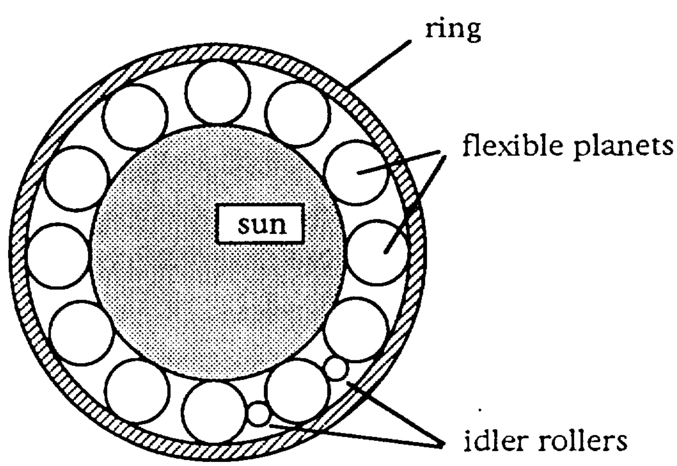


Figure 1: Roll Ring Device

The flexible planets of a Roll Ring device eventually run into one another by *walking* (a result of microslip) or *jerking* out of position. One solution has been to run only one planet per Roll Ring device, drastically reducing its current carrying capability. Another

has been to add idler rollers between the planets as shown in Figure 1. This solution increases the complexity of the device, while preventing its future reduction in size for use in electric motors. The objective of this research is to find flexible planets that can transfer electricity while maintaining their own positions in the annulus.

Original Flex-Gears:

The original Flex-Gear device was conceived as a standard planetary gear train whose planets maintain their own position by standard gear meshing. Hollowed for additional flexibility, Flex-Gears are planet gears that are compressed in the annulus of their sun and ring gears like the planets of a Roll Ring device to maintain electrical contact.

Planet gears offer more regions of contact with their sun and ring gears than the planets of a Roll Ring device through multiple tooth contact. Adjacent teeth on most gears simultaneously touch the mating gear throughout part of gear rotation to share mechanical load. This multiple tooth contact is reflected by the contact ratio;

$$cr = \text{arc of tooth action} / p \qquad (1)$$

where the arc of tooth action is shown in Figure 3a as the length along the pitch circle in the sum of the angles of approach and recess. The circular pitch p is the length along the pitch circle between the corresponding points of two adjacent gear teeth.

Contact ratios of gears with a 20° pressure angle can range from 1.0 to 1.97 [Cowie, 1961]. A contact ratio of 1.9 promises dual tooth contact for 90% of gear rotation and single tooth contact throughout the remaining rotation. Using many flexible planet gears per Flex-Gear device achieves an average of 90% more contact regions than a the Roll Ring device. The number of contact regions, however, does not increase electrical flow as much as contact area and pressure do.

Increasing the electrical contact of Flex-Gears by adding contact pressure is accomplished simply by increasing the compression of the Flex-Gear in the annulus of its sun and ring gears. However, the

additional pressure on standard gear teeth would promote significant wear and power loss, since gear teeth slide on their mating teeth upon meshing. Increasing the electrical contact of Flex-Gears by adding contact area requires significant gear tooth deformation. This deformation would hinder proper gear meshing, further increasing sliding between mating teeth.

Although maintaining their own position in the annulus of their sun and ring gears, Flex-Gears are predicted to make poor electrical contact due to their low contact area and pressure or incur severe wear and power loss. Roll Rings make sound electrical contact, but cannot maintain their position without the help idler rollers.

Rolling-Flex-Gears:

Combining the advantages of both Roll Rings and original Flex-Gears, *Rolling-Flex-Gears* are similarly compressed in the annulus of their sun and ring gears. Rolling-Flex-Gears attempt to rid sliding from the surfaces that transfer electricity and improve electrical contact by using two entirely different types of contact surfaces: one to transfer electricity and the other to maintain the position of the gear. The surfaces that position the Rolling-Flex-Gear, called *positioning surfaces*, will not wear severely by sliding since they conduct little or no electricity. The surfaces of electrical contact should be flexible and forced against their mating surfaces to produce a sufficient contact area and pressure for electrical flow. To minimize wear, the electrical contact surfaces should roll rather than slip and are therefore called *rolling surfaces*.

This brings the design back to Roll Rings whose surfaces offer purely rolling electrical contact. Adding spurs to the flexible rings in the Roll Ring device makes them into Rolling-Flex-Gears that reposition themselves upon slipping out of position. If spurs are added to the planets, corresponding valleys must be added to the ring and sun and visa versa. If the entire Rolling-Flex-Gear is made of a metal, electricity may flow through the spurs. Nevertheless, the rolling surfaces should transfer most of the electricity with their high pressure and contact area.

Figure 2 shows a Rolling-Flex-Gear with straight valleys and its sun or ring with semi-circular top spurs. As can be seen from this Figure, the Rolling-Flex-Gear can slide freely between the spurs. Held in position by the friction between the rolling surfaces, the Rolling-Flex-Gear is designed to roll over the spurs without making contact. Should slipping, however, occur by walking or jerking out of position, the spurs will reposition the Rolling-Flex-Gear.

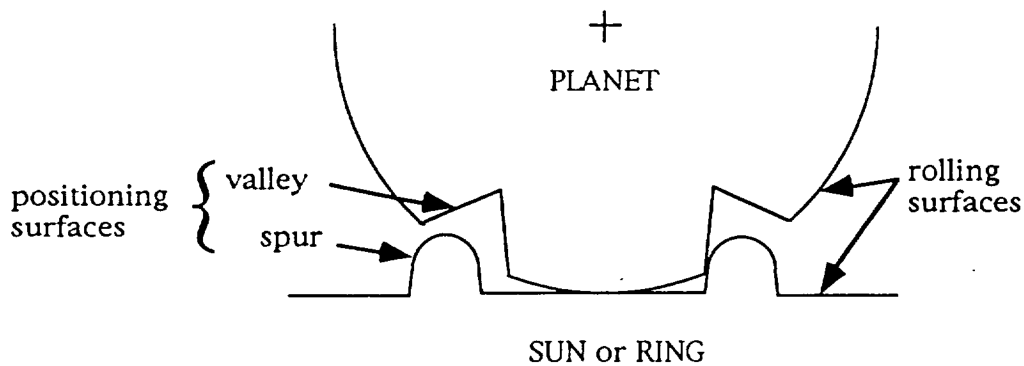


Figure 2: A Rolling-Flex-Gear

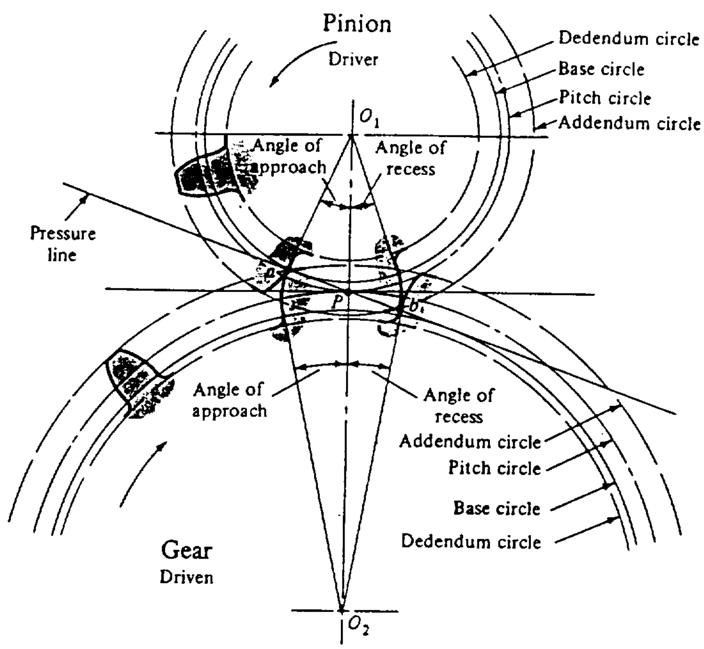
Upon repositioning, sliding occurs between both rolling and positioning surfaces. Nevertheless, it is the sliding on the rolling surfaces that especially contributes to wear and power loss because of their high contact pressure. To diminish wear and power loss, a search began for spur and valley shapes that would minimize sliding on the rolling surfaces. Candidate spur curves included simple shapes as shown in Figure 2 and those common to gear technology, such as, cycloids and involutes. Cycloid curves seemed to be natural solutions since they are generated by rolling discs. However, it was an innovative choice of involute curves that reduced the sliding on the rolling surfaces to zero.

Pitch-Rolling-Gears:

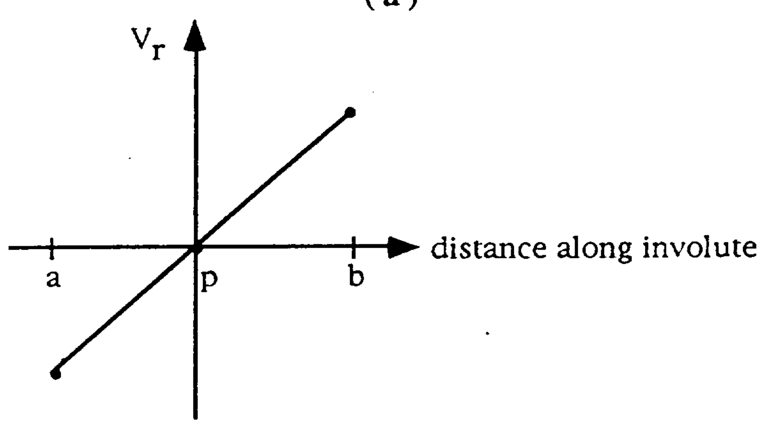
This section of the report will explain the development of Pitch-Rolling-Gears by considering rigid gears; the next section will explain the addition of flexibility.

Standard involute gear meshing is shown in Figure 3a. Initial contact is made at point *a*. A combination of sliding and rolling

occurs between the mating teeth as the point of contact follows the *pressure line* until separation at point *b*. The section of the pressure line between points *a* and *b* is called the *line of action*. The *dedendum circle* marks the roots of the gear teeth, while the *addendum circle* marks their tops. The *base circle* marks the diameter from which involutes are drawn to form gear teeth. Base circles lay tangent to the pressure line.



(a)



(b)

Figure 3: (a) Standard Gear Meshing [Courtesy of Shigley, 1989]
(b) Standard Gear Tooth Sliding

It is the relative velocity of mating teeth at their point of instantaneous contact that produces tooth sliding. The component of relative velocity that is tangent to the surfaces of contact is graphed in Figure 3b. Notice that the tangential relative velocity is zero at the pitch point p, indicating pure rolling at that point. This says that an involute gear rotates as if it were rolling on the imaginary *pitch circle* of its mating gear. The transfer of constant angular velocity is a characteristic of involute gears called *conjugate action.* Pitch-Rolling-Gears take advantage of this to eliminate sliding on their surfaces of electrical contact.

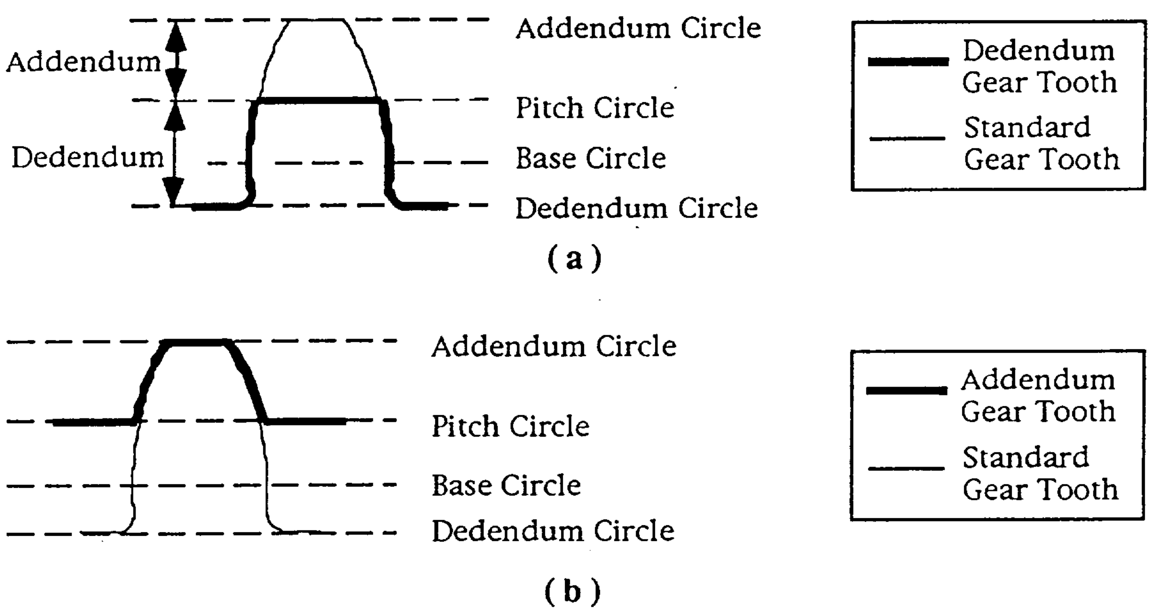
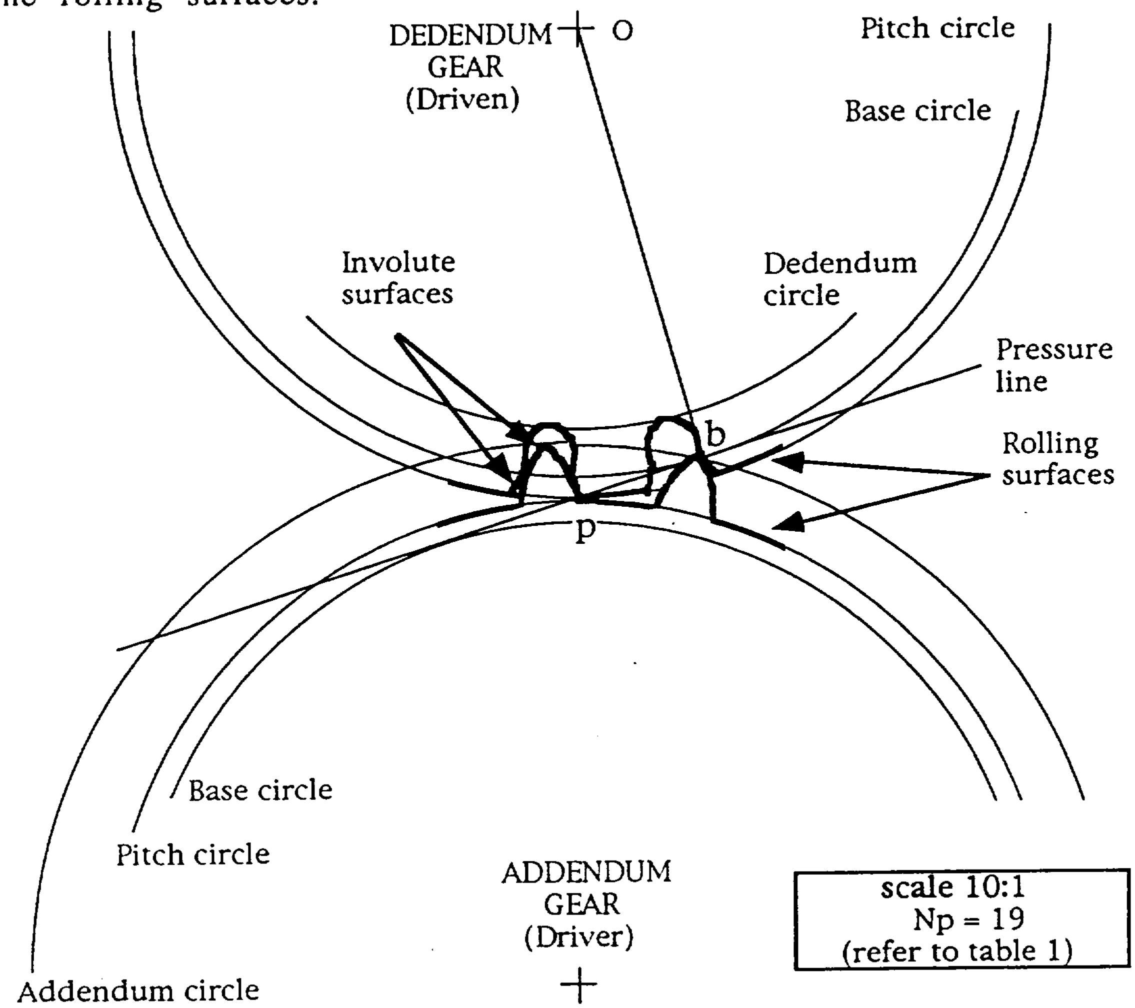


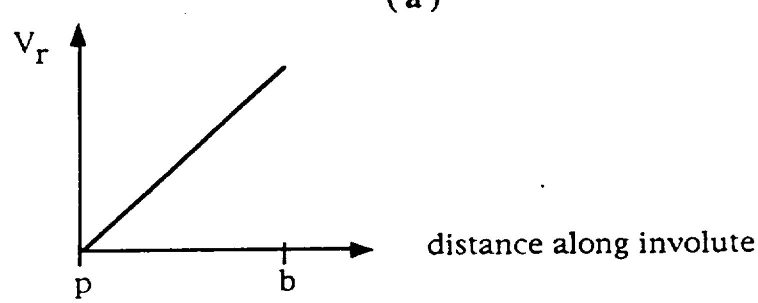
Figure 4: (a) Dedendum Gear Tooth (b) Addendum Gear Tooth

This advantage is implemented by the formation of special teeth that are shown in Figure 4. A *dedendum gear tooth* is formed by cutting off the addendum of a standard gear tooth. An *addendum gear tooth* is formed by filling in its dedendum. Upon meshing, the cut off portions of a *dedendum gear* roll on the filled in portions of the *addendum gear* to produce *rolling contact.* These surfaces are called *rolling surfaces* like those of a Rolling-Flex-Gear. Meanwhile, gear contact along the involutes maintains the relative positions of

the gears and is referred to as *involute contact.* The *involute surfaces* are special positioning surfaces that maintain pure rolling on the rolling surfaces.



(a)



(b)

Figure 5: (a) Pitch-Rolling-Gear Meshing
(b) Pitch-Rolling-Gear Tooth Sliding

In Figure 5a, the dedendum gear teeth have been widened and addendum gear teeth slimed to maximize the rolling surface area. The leftmost addendum gear tooth makes initial contact with the dedendum gear at the pitch point p. Sliding and rolling occurs between the mating teeth along the line of action until separation at point b, similar to standard gear meshing.

To increase the rolling surface area, Pitch-Rolling-Gear teeth are spaced as far apart as possible. Standard gear teeth are spaced more closely so that adjacent teeth share mechanical loads. The resulting contact ratios of standard gears are commonly between 1.2 and 1.7. Since Pitch-Rolling-Gears carry little mechanical load, a contact ratio of 1.0 is used for maximum tooth spacing. Gear tooth spacing can be further increased by extending the length of the line of action. For a given pressure angle ϕ, its length depends on the height of the addendum gear teeth or the diameter of the addendum circle. Referring to Figure 3a, its maximum length occurs when the addendum circle passes through the intersection of the pressure line and the pitch circle of the dedendum gear. Further extension would force the contact ratio to drop below 1.0, thereby allowing catastrophic slipping between gears.

Extending the addendum circle past the tangent point of the pressure line and the base circle causes the addendum teeth to dig into the dedendum teeth. This *mechanical interference* between mating gear teeth occurs inside the base circle of the dedendum gear. Interference is routinely remedied in standard gears by *undercutting* the gear teeth involved. Undercutting removes the material from the gear teeth where interference occurs. Pitch-Rolling-Gears mesh such that only the dedendum gear teeth require an undercut. Since the dedendum gear teeth have already been widened, undercutting has little effect on the strength of dedendum teeth. However, the further the addendum circle is extended, the more total sliding occurs between mating teeth. Hence, the cost of increasing gear tooth spacing or rolling surface area is the increase in sliding between teeth. Until this trade-off can be quantified, Pitch-Rolling-Gears will assume a maximum length in their lines of action without requiring an undercut.

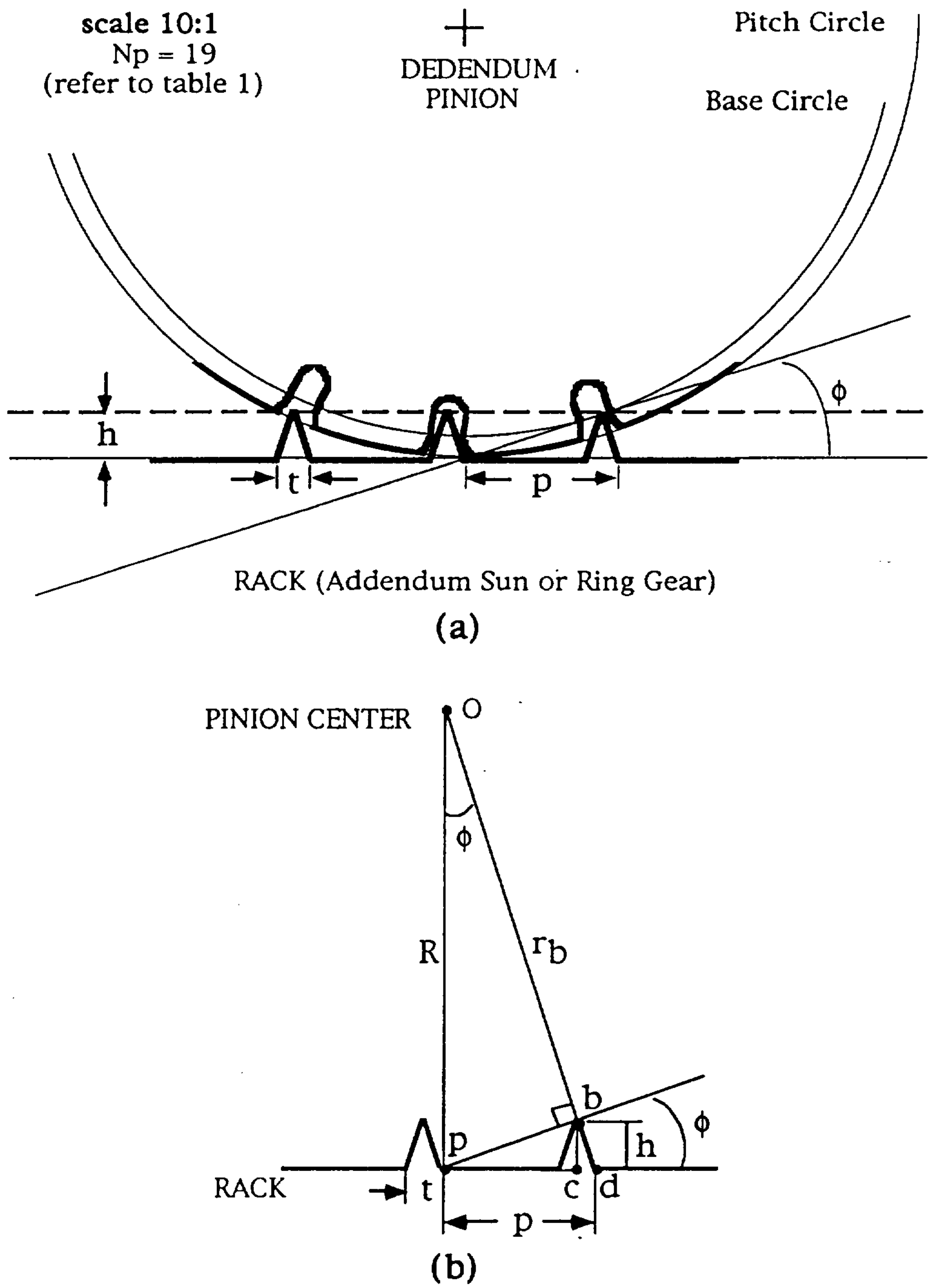


Figure 6: (a) Rack and Pinion (b) Rack and Pinion Geometry

Total sliding is realized by integrating the relative velocity along the involute. Comparison of Figures 3b and 5b indicates that for equal contact ratios and pitch diameters, Pitch-Rolling-Gears slide

more than standard gears. The additional sliding on Pitch-Rolling-Gears is the cost of pure rolling on their surfaces of electrical contact. Nevertheless, severe wear between the involute surfaces of Pitch-Rolling-Gears should not result, since there is little pressure or flow of electricity between these surfaces.

Planet gears, which roll in the annulus of their sun and ring gears, can be addendum or dedendum gears. Since addendum gears mesh with dedendum gears only, the choice of the planet gear type determines the sun and ring gear type. This choice does not affect the rolling surface area of a single planet gear but can affect the total rolling surface area of an entire device. As shown in Figure 4, addendum gear teeth lie outside the pitch circle, while dedendum gear teeth are enclosed by it. So, for equal pitch diameters, a dedendum gear has a smaller outside diameter, equal to its pitch diameter, than an addendum gear that has an outside diameter equal to that of its addendum circle. With their smaller outside diameters, dedendum gears can be packed more closely in an annulus to provide more contact area.

Thirty 1/2" diameter dedendum planet gears fit in the annulus of a five inch diameter sun gear and a six inch ring gear. Since each planet is over ten times smaller than its sun and ring gears, the gear meshing geometry can be approximated by the rack and pinion shown in Figure 6a. Having chosen a contact ratio of 1.0 and the longest line of action possible without introducing tooth interference, the corresponding geometry is shown in Figure 6b.

By definition, the circular pitch *p* is given by,

$$p = 2\,\pi\,R/N_p \qquad (2)$$

and indicates the distance between teeth along the pitch circle, as shown in Figure 6b.

By triangle opd, the pressure angle ϕ is given by,

$$\tan\phi = P/R \qquad (3)$$

From triangle opb, the radius of the base circle of the pinion is given by,

$$r_b = R \cos \phi \tag{4.1}$$

Then the rack tooth height h can be found from,

$$h = R - r_b \cos \phi \tag{4.2}$$

Eliminating r_b from Equations 4.1 and 4.2 and simplifying,

$$h = R \sin^2\phi \tag{4.3}$$

Points o,b, and d lie on a straight line such that the thickness of a rack tooth is related to its height h by,

$$t = 2\ h \tan \phi \tag{5}$$

To compare the rolling surface area of different Pitch-Rolling-Gears, a characteristic number λ is defined as the ratio of the Pitch-Rolling-Gear surface area to that of a flexible ring of the same pitch diameter. So,

$$\lambda = N_p\ (p-t)/2\ \pi\ R = 1 - t/p \tag{6}$$

Table 1 shows the specifications of several 1/2" diameter Pitch-Rolling-Gears. Choosing the number of planet teeth N_p, Equations 2 through 6 were applied consecutively to find p, ϕ, h, t, and λ. As N_p increases in table 1, tooth height h and thickness t decrease, while the characteristic number λ increases. This demonstrates that electrical contact increases with decreasing tooth size. In fact, Equation 6 shows that zero dimension teeth yield the maximum electrical contact of λ=100%. This is intuitively obvious since zero dimension teeth is the case of a plain ring on a toothless surface. Nevertheless, finite teeth are required to maintain the positions of the Pitch-Rolling-Gears with respect to their sun and ring gears.

The minimum size of the teeth on the sun and ring gears will be determined by the tooth strength required. The sun and ring

teeth must be strong enough to withstand the forces that arise from maintaining the position of each planet. By neglecting inertial forces, it can be shown that zero force arises in a *perfectly* manufactured planetary gear train. However, manufacturing error causes some planet sliding in the annulus. This sliding results in friction forces between the rolling surfaces that must be overcome by the gear forces in order to maintain the position of each planet. Standard pressure angles such as 14.5° and 20° will be attempted by varying the Pitch-Rolling-Gear radius R.

Name	Symbol	Equation			
Number of Planet Teeth	N_p	--	18	19	20
Circular Pitch (inches)	p	2	.087	.083	.079
Pressure Angle (degrees)	ϕ	3	19.2	18.3	17.4
Tooth Height (inches)	h	4.3	.027	.025	.023
Tooth Thickness (inches)	t	5	.019	.016	.014
Characteristic Contact	λ	6	.78	.80	.82

Table 1: Flex-Gear Specifications

Adding Flexibility to Pitch-Rolling-Gears:

The current carrying capability of a Pitch-Rolling-Gear depends greatly on the contact area between rolling surfaces. Contact area can be increased by adding flexibility to the rolling surfaces of either the planet gears or sun and ring gears or both. Ring gears are hollow and need only sufficiently thin walls to be flexible enough for electrical contact. Sun and planet gears can be hollowed to achieve the same effect.

To determine the mechanical possibility of hollow planet gears, consider a planet without any teeth, as shown in Figure 7. Compressed in an annulus with 0.005 inches of nominal interference, this flexible ring is designed to maintain contact assuming an annulus tolerance of ±0.005 inches. The tolerance of the flexible ring is not considered.

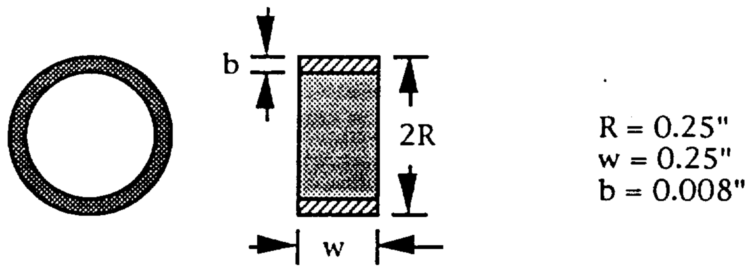


Figure 7: A Flexible Ring or Toothless Flex-Gear

As discussed in the Appendix, the sensitivity of ring stress to the inaccuracies of the annulus within standard machining tolerances is high. Since the addition of tooth corrugations would further increase these stresses, hollow planet gears with pitch diameters around 0.5" seems unfeasible. Completely flexible Pitch-Rolling-Gears, whether they are sun, ring, or planet, may upset involute meshing. This upset could inhibit conjugate action to introduce more sliding on the rolling surfaces.

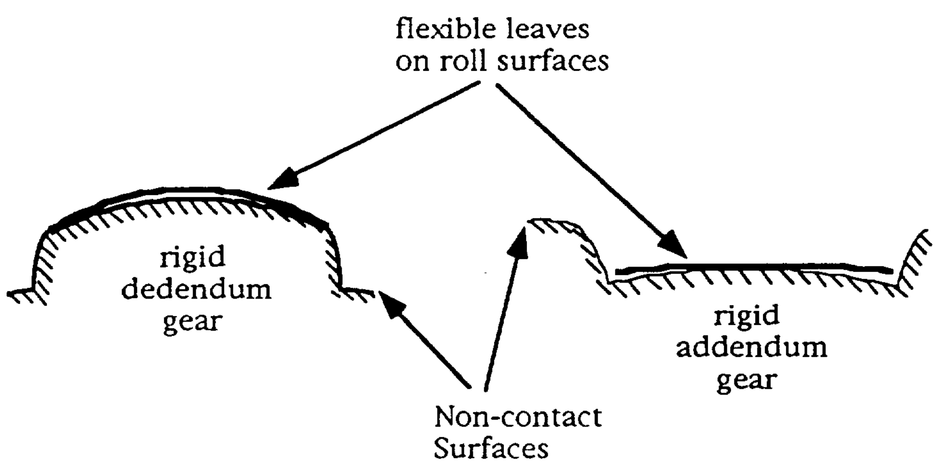


Figure 8: Semi-rigid Pitch-Rolling-Gears

Semi-rigid Pitch-Rolling-Gears are favored, since they incur much less stress than hollow gears and maintain their conjugate action. For increased area of electrical contact, the rolling surfaces must be flexible. However, to maintain conjugate action between the gears, the involute surfaces should be rigid, thus forming a semi-rigid Pitch-Rolling-Gear. Figure 8 shows an example of both

dedendum and addendum semi-rigid Pitch-Rolling-Gears. Flexibility has been added only to the rolling surfaces with flexible leaves. In this way flexibility can be added to the planet gears or sun and ring gears or both.

Consider the electrical effectiveness of a Pitch-Rolling-Gear planet whose flexibility has been added by either hollowing or the addition of flexible leaves. Assume that when the planet contacts its sun and ring gears on its rolling surfaces, its electrical contact is the same as that of the flexible ring shown in Figure 7. Made of standard casting Beryllium Copper*, this flexible ring has a contact area of approximately 0.018 square inches with its sun and ring (see Appendix). Assuming a 200 A/in^2 maximum current density across the interfaces [Still, 1916], 3.5 A may be transferred through a single flexible ring.

Because contact is lost when the planet straddles a tooth, its average current capability is reduced according to the size of the sun and ring gear teeth. As a starting design Figure, consider sun and ring teeth of 0.025 inches high (the approximate tooth size of a 96 diametral pitch standard gear). According to Equation 6, the corresponding planet gear makes electrical contact through its rolling surfaces for 80% of its rotation in the annulus. Since each planet in the device straddles a tooth at different positions of the sun gear, this results in an average 20% loss of current carrying capability for the entire device. Hence, a flexible Pitch-Rolling-Gear planet will conduct up to 2.8 Amperes for an average of up to 80 Amperes for an entire device with 30 flexible planets.

Potential Problem Areas:

In addition to the loss of electrical contact, two potential problems arise when a planet straddles a tooth on its sun or ring gear. First, the planet loses its center position in the annulus, resulting in a loss of conjugate action that introduces sliding on the rolling surfaces. However, the wear may not be severe since there is little contact or normal force at this region of the rolling surface.

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* specific OSHA requirements exist for manufacture (BeCu is toxic).

Second, the planet impacts the sun and ring as each tooth touches down for electrical contact. This impact will not be severe at low speeds, and may be prevented entirely at high speeds by the inertia of the Flex-Gear.

A very small amount of sliding on the rolling surfaces may also arise from manufacturing error in the roll surface diameters of the ring, sun, and planet gears.

Future Work:

A. Evaluate the friction, normal, and gear forces that occur in a flexible Pitch-Rolling-Gear device and size its gear teeth accordingly.

B. Investigate the electrical contact resistances of various material combinations, accounting for surface pressure and smoothness to more accurately predict the current carrying capability of flexible Pitch-Rolling-Gears.

C. Consider the extreme temperature effects of space and the use of dissimilar metals.

D. Using the information gained from steps A, B, and C, design a flexible Pitch-Rolling-Gear device for prototype. Consider using Pitch-Rolling-Gears with undercut.

E. Quantify the mechanical efficiency (energy loss) of Pitch-Rolling-Gears.

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Appendix: Flexible Ring Calculations

Consider the flexible ring, shown in Figure A1, compressed in the annulus of a sun and ring like the Roll Rings described in the *Introduction* of this report. Assume that subsequent loading and deformation occurs as shown in Figure A.2.

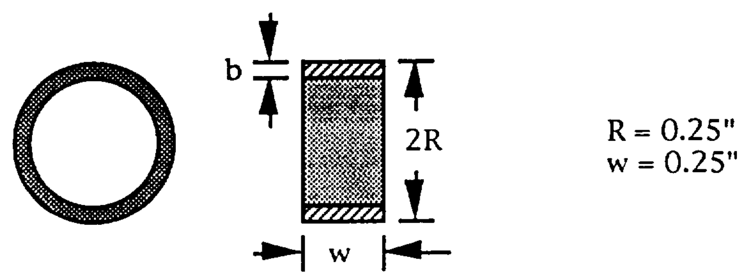

Figure A.1: A Flexible Ring or Toothless Flex-Gear

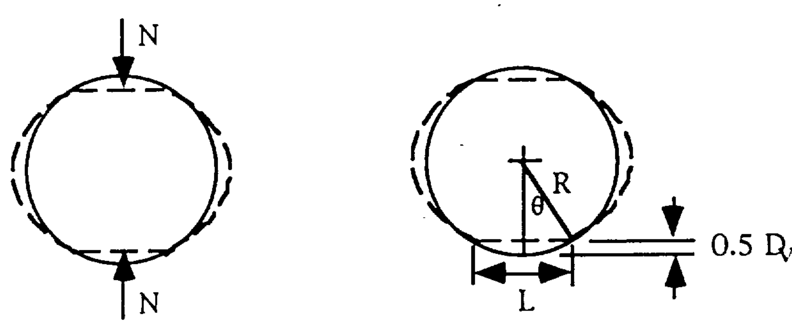

Figure A.2: Flexible Ring Loading and Deformation

Approximating the flexible ring as a curved beam, its circumferential stress due to bending is

$$\sigma = M\, c\, /\, I \qquad \text{(A.1)}$$

where *M* is the internal moment of the ring and *I* is the area moment of inertia of the cross section of the ring. For a flexible ring with the rectangular cross section shown in Figure A.1, $c=b/2$ and $I=w\, b^3/12$. According to Young [1989], the maximum internal moment *M* occurs at the point loads shown in Figure A.2 and is

$$M = N\, R/\pi \qquad \text{(A.2)}$$

where N is the normal force shown in Figure A.2 and is given by

$$N = - D_V \, E \, I \, / \, (0.149 \, R^3) \qquad \text{(A.3)}$$

D_V is the vertical deflection of the flexible ring, as shown in Figure A.2. The flexible ring is to be compressed in an annulus with 0.005 inches of nominal interference and is designed to maintain contact assuming an annulus tolerance of ±0.005 inches. Neglecting the tolerance of the flexible ring the vertical deflection varies from .000" to .010".

Eliminating I, M, and N from Equations A.1, A.2, and A.3,

$$\sigma = - D_V \, E \, b \, / \, 2 \, (0.149 \, \pi \, R^2) \qquad \text{(A.4)}$$

This equation shows that the ring stress σ is proportional to the vertical deflection D_V and ring thickness b, while inversely proportional to the planet radius R. This means that changes in the vertical deflection, which are due to manufacture error, result in proportional variations in ring stress. This sensitivity can be significantly offset by increasing the planet radius R.

Find the maximum ring thickness b

Consider the flexible ring in Figure A.1 to be made from copper alloy C82500. This alloy, called standard casting Beryllium Copper, is made from 97.2%Cu, 2%Be, 0.5%Co, 0.25%Si* [American Society for Metals, 1978] and has the following properties:

E (elastic modulus= 18.5×10^6
σ_e (endurance limit for 5×10^7 cycles) = 24 ksi

Using σ_e as the maximum allowable stress and D_V=.010" and R=.25" in Equation A.4,

$$\boxed{b = .008"}$$

* Si (Silicone) adds strength, reduces conductivity, and may cause noise.

Find the average contact area for the flow of electricity:

The average contact area for the flow of electricity is the average contact area that the flexible ring has with its sun and ring. Figure A.2 shows that the circumferential length of contact L is given by

$$L = 2 R \sin\theta \qquad (A.5)$$

and the vertical deflection is

$$D_v = 2 R (1 - \cos\theta) \qquad (A.6)$$

Eliminating θ from Equations A.5 and A.6 and substituting in Dv(avg)=.005" and R=.25", yields

$$L_{avg} = .0705"$$

Multiplying the average circumferential length of contact L_{avg} by the flexible ring width w, the contact area of the flexible ring with its sun and ring is found:

$$\boxed{\text{contact area} = .018 \text{ in}^2}$$

Find the average contact pressure for the flow of electricity:

The average contact pressure for the flow of electricity is that which is made by the flexible ring on its sun or ring:

$$P_{avg} = N_{avg} / w\, L_{avg} \qquad (A.7)$$

where the average normal force N_{avg} and contact length L_{avg} are evaluated by using the average vertical deflection $D_{v(avg)}$=.005". Substituting N_{avg} =0.42 lbs, w=.25", and L_{avg}=.0705" into Equation A.7,

$$\boxed{P_{avg} = 24 \text{ psi}}$$

This contact pressure is fifteen times higher than what is considered sufficient for good electrical contact between a brush and ring [Still, 1916].